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Improving the Reliability of a Reciprocating Compressor for Applications in a Refrigerator

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Abstract To enhance the reliability of a newly designed reciprocating compressor applied in a domestic compressor, accelerated life tests were developed using new definitions of the sample size and the B_1 life index. In 1^{st} accelerated life testing, the compressor was locked due to the fracture of the suction reed valve. The failure modes and mechanisms of the suction reed valve in the accelerated tests were found to be similar to that of the failed product in the field. The root cause of the failure was the overlap between the suction reed valve and the valve plate in the suction port. The missing parameters in the design phase were modified by expanding the trespan size, introducing tumbling process, changing the material and thickness for the valve, introducing a ball peening and brushing process for the valve plate. In 2^{nd} accelerated life testing, the compressor was locked due to the interference between the crank shaft and thrust washer. The corrective plan was to heat treat the crank shaft. The B_1 life of the compressor improved from 1.5 to 12.9 years.

Key words Reciprocating compressor, Reliability design, Load analysis, Accelerated life testing

1. Introduction

Most compressor companies are making every effort to develop more efficient, high-volumetric compressors. At the same time, there are market forces requiring cost reductions of the product, which leads companies to seek cheaper and more reliable parts. As attempts are made to provide a newly designed compressor, reliability needs to be evaluated by a proper testing method.

Under typical customer usage conditions, the compressor is very important for achieving consistent operation over a specified period. It can be designed by selecting the optimum variables such as signal factors. When these factors are fixed at the right levels, they also will make the compressor functions “robust”, that is, insensitive to noise factors.[1)] However, some factors in the design review often have been neglected, and the more it is hard to expect the part life and failure rate from the testing results. In the marketplace, these minor design flaws may shorten the product reliability.

Preventing the design flaws of the new product is an important factor in the product development process of design, production, shipping and field testing. Conventional methods, such as product inspection, rarely reproduce the reliability problems occurring in marketplace use. Designing for the maximum reliability requires extensive reliability testing at each development step. As a result, the cost of quality assurance and appraisal may increase significantly.

In reliability testing, many product designers focus on the accelerated life testing (ALT) method. This method can help shorten the product development cycles, costs less money, and clarify diverse design faults. However, there are some caveats of using ALT: any failures after ALT may not represent those occurring in market conditions. This problem usually arises because of the inconsistency of the direction and magnitude of the load, such as force or pressure in system dynamics. Moreover, the number of test samples and the test times may be insufficient to uncover unusual failure modes. ALT should be performed with sufficient samples and test time. ALT equipment can and should be designed to match product loads.

When rotary compressors were abnormally failing in 1987, there were massive recalls of the compressor. Because oil sludge, formed in the refrigeration cycle, blocked the capillary tubes, the function of the refri-

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gerator was lost. In the newly designed compressor, preventing a failure mode such as a blocking of the compressor was very important to the reliability of the refrigerator. However, reliability testing methods such as ALT were not used extensively at the time.

To ensure the reliability of newly designed compressors, new methodologies for ALT the theoretical background of a new B_X life were introduced.[2] The procedures for the compressor reliability design can be summarized as (1)load analysis of the refrigeration cycle; (2) fabrication of ALT equipment; and (3) performance of several ALT tests to predict reliability. For the loads of the compressor, it is important to use a parameter design, the traditional thermodynamic cycle model,[3-5] 2nd refrigerant properties at each state.[6] ALT equipment can also be fabricated on the basis of load analysis.

In this paper, the reliability design of a compressor is investigated with new ALT methodologies. The pressure conditions in a vapor-compression cycles using mass conservation and energy conservation are first characterized. After a sequence of ALTs, we evaluated the compressor reliability with Bx life. Finally, the effectiveness of these methodologies for the reliability design of the newly designed compressor is demonstrated.

2. Theoretical background

2.1 Field Application problems

The suction reed valves of domestic refrigerator compressors used in the field were cracking and fracturing, leading to failure of the valve (Fig. 1). Specific customer usage conditions and patterns leading to the failures were unknown. Because the compressor would lock up when the valve failed, the function of refrigerator was lost and customers would ask to have the refrigerator replaced. To solve the problem, it was very important to reproduce the field failure mode of the suction reed valve in the laboratory.

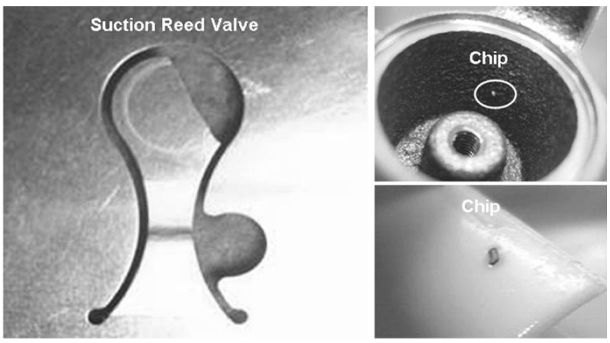


Fig. 1. Fracture of the compressor suction reed valve in the field.

The failed compressor in the market place might have had two structural design flaws: (1) Suction reed valve had an overlap with the valve plate; and (2) The valve plate had a sharp edge. When the suction reed valve impacted the valve plate continually, it would fracture easily.

2.2 Load analysis of compressor

A refrigerator consists of a compressor, a condenser, a capillary tube and an evaporator. The vapor compression refrigeration cycle receives work from the compressor and transfers heat from the evaporator to the condenser. The main function of the refrigerator is to provide cold air from the evaporator to the freezer and refrigerator compartments.

A capillary tube controls flow in the refrigeration systems and drops the high pressure of the refrigerant in the condenser to the low pressure in the evaporator. In a refrigeration cycle design, it is necessary to determine both the condensing pressure, Pc, and evaporating pressure, Pe. These pressures depend on ambient conditions, customer usage conditions, and heat exchanger capacity in the initial design stage (Fig. 2).

The mass flow rate of refrigerant in a compressor can be modeled as

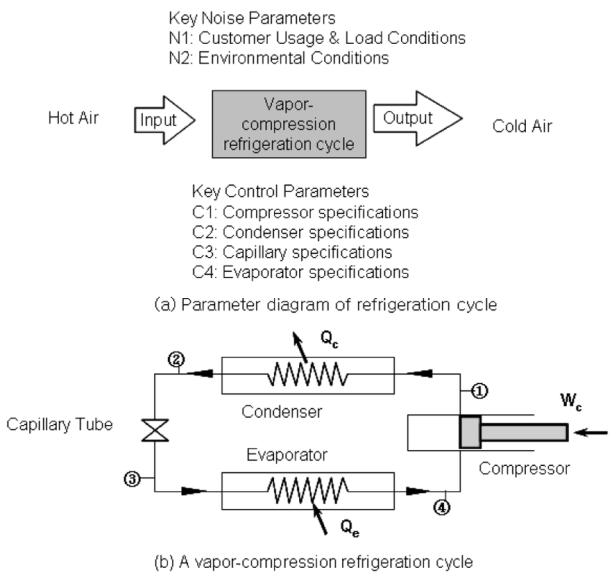


Fig. 2. Schematic diagram for a vapor compression cycle: (a) Parameter diagram of refrigeration cycle, (b) A vapor-compression refrigeration cycle.

$$\dot{m} = PD \times \frac{\eta_v}{v_{suc}} \tag{1}$$

where PD is the volume flow rate, η_v is the volumetric efficiency, and is the specific volume. The mass flow rate of refrigerant in a capillary tube can be modeled as[7]

$$\dot{m}_c = A\left[\frac{-\int_{P_2}^{P_3} \rho\, dP}{\frac{2}{D} f_m (L_3 - L_2) + 1n\left(\frac{\rho_2}{\rho_3}\right)}\right]^{0.5} \tag{2}$$

where A is the cross area of the capillary tube, ρ is the refrigerant density, f_m is the mean friction coefficient, L_3 - L_2 is the capillary length of the two-phase interval, η_v is the volumetric efficiency, and is the specific volume.

By conservation of mass, the mass flow rate can be determined as,

$$\dot{m} = \dot{m}_c \tag{3}$$

The heat transfer in the condenser can be described as

$$Q_c = \dot{m}(h_1 - h_2) = (T_c - T_o)/R_c \tag{4}$$

The heat transfer in the evaporator can be described as

$$Q_c = \dot{m}(h_4 - h_3) = (T_i - T_e)/R_e \tag{5}$$

When nonlinear Eqs. (3), (4) and (5) are solved, the mass flow rate, $\dot{m}$, evaporator temperature, T_e, and condenser temperature, T_c, can be determined. Since the saturation pressure, P_{sat}, is a the function of temperature, the evaporator pressure, P_e (or condenser pressure P_c), can be obtained as:

$$P_e = f(T_c) \tag{6}$$

The internal stress of the compressor depends on the pressure difference suction pressure, P_{suc}, and discharge pressure, P_{dis}. That is,

$$\Delta P = P_{dis} - P_{suc} \cong P_c - P_e \tag{7}$$

By repeating the on and off cycles, the compressor receives the dominant stress. Under accelerated stress conditions, the life-stress model (LS model)[8] can be modified as

$$T_f = A(S)^{-n} \exp\frac{E_a}{kT} = A(\Delta P)^{-n} \exp\frac{E_a}{kT} \tag{8}$$

where A is constant, T_f is the time to failure, k is Boltzman's constant, E is the activation energy, T is the absolute temperature and n is the quotient. So the acceleration factor (AF) can be derived as

$$AF = \left(\frac{S_1}{S_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right]$$

$$\left(\frac{\Delta P_1}{\Delta P_0}\right)^n \left[\frac{E_a}{k}\left(\frac{1}{T_0} - \frac{1}{T_1}\right)\right] \tag{9}$$

where S_1 (or P_1) is mechanical stress (or pressure difference) under accelerated stress conditions, and S_0 (or P_0) is mechanical stress (or pressure difference) under normal stress conditions.

2.3 Theoretical background of new definition of Bx life for the accelerated life test

Traditionally, the characteristic life is defined as[9]

$$\eta^\beta = \frac{\sum t_i^\beta}{r} \cong \frac{n \cdot h^\beta}{r} \tag{10}$$

where β is the shape parameter in a Weibull distribution.

As product (or part) reliability improves, failure of the part becomes less frequent in laboratory tests. It becomes more difficult to evaluate the characteristic life in Equation (10). When the failed sample number is below four, it follows the Poisson distribution.[2] For a 60 percent confidence level, the characteristic life can be redefined as

$$\eta^\beta \cong \frac{1}{r+1} \cdot n \cdot h^\beta \tag{11}$$

In order to introduce the B_X life in the Weibull distribution, the characteristic life can be modified as

$$L_B^\beta \cong x \cdot \eta^\beta = \frac{x}{r+1} \cdot n \cdot h^\beta \tag{12}$$

where L_B = B_X life and x = 0.01X, on the condition that $x \leq 0.2$.

B_X is the time at which X% of the compressors installed in a particular total population of refrigerators will have failed over a specified time.

In order to assess the B_X life with about a 60 percent confidence level, the number of test samples is derived in Equation (12). That is,

$$n \cong \frac{1}{x} \cdot (r+1) \cdot \left(\frac{1}{h^*}\right)^\beta \tag{13}$$

on the condition that the durability target, $h^* = h/L_B \geq 1$.

3. Experimental Procedure

The compressor was subjected to ambient conditions from 0 to 50°C and 0 to 85% relative humidity, and vibrations from 0.2 to 0.24 g. The system was subjected to 22 on-off cycles per day under normal operating conditions. A worse case was also simulated with 98 on-off cycles per day. Under the worst case conditions, the compressor operation for 10 years would be 357,700 cycles (Table 1).

From the test data of the worst case, normal pressure was 1.27 MPa and the compressor dome temperature was 90°C. For accelerated life testing, the acceleration factor (*AF*) for pressure was 2.94 MPa and the compressor dome temperature was 120°C. With a quotient, n, of 2, total *AF* was calculated from Eq. (8) and was 20.9 (Table 2).

The test cycles and the number of samples used in ALT were calculated as follows:

$$n \cong (r+1)\cdot\frac{1}{x}\cdot\left(\frac{L_B}{AF\cdot h}\right)^{\beta} \quad (14)$$

where *r* is failed numbers; *n* is the test sample numbers; *x* is 0.01; *AF* is the acceleration factor; *h* is testing cycles; and L_B is the target B_X life. If the shape parameter is 1.9, the test cycles and test sample numbers calculated in Eq. (14) were 40,000 cycles and 20EA, respectively. The ALT was designed to assure a B_1 of 10 years life with about a 60 percent level of confidence if no unit fails during 40,000 cycles.[2)]

For the ALT experiments, a simplified vapor compression refrigeration cycle was fabricated. It consisted of an evaporator, compressor, condenser, and capillary tube. A fan and two 60-W lamps maintained the temperature within the insulated (fiberglass) box. A thermal switch attached on the compressor top controlled a 51 m3/h axial fan. The test conditions and test limits were set up on the control board. As the test began, the high-side and low-side pressures could be observed on the pressure gauge or display monitor (Fig. 3).

Fig. 4 shows the P-*h* diagram and the duty cycles of pressure difference suction pressure, P_{suc}, and discharge pressure, P_{dis}.

Table 1. Operating number of a reciprocating compressor

Item	Operating cycle(times)			
	1 day		10 years	
	Normal	Worst	Normal	Worst
Compressor	22	98	80,300	357,700

Table 2. ALT conditions in a vapor compression cycles

System conditions		Worst case	ALT	AF
Pressure, kg/cm2	High side	13.0	30.0	5.3
	Low side	0.0	0.0	
	P	13	30	
Temp., °C	Dome Temp.	90	120	3.9
Total *AF*		-		20.9

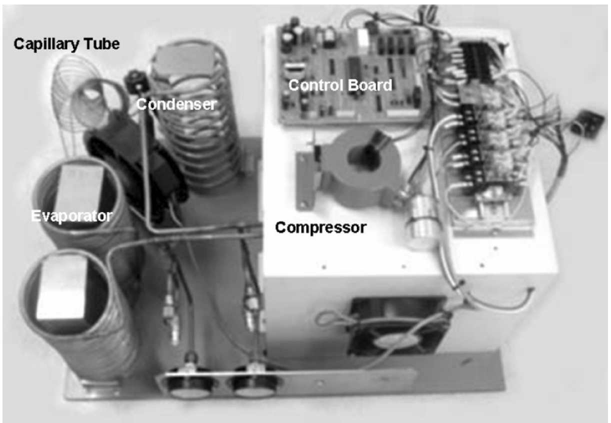
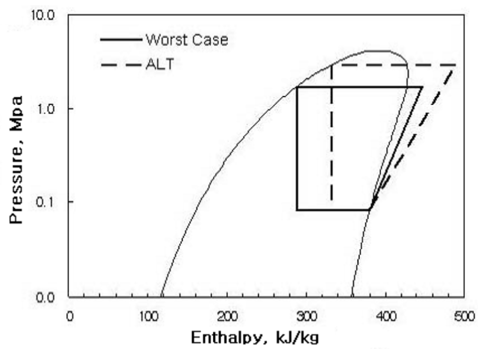
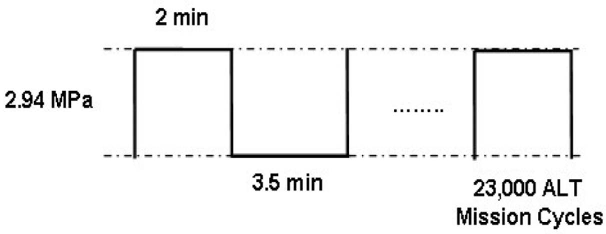


Fig. 3. Equipment for the accelerated life tests.



(a) P-*h* diagram



(b)Duty cycles on the ALT equipments

Fig. 4. *P*- *h* diagram and duty cycles.

4. Results and discussion

4.1 Validity of the accelerated life test and failure analysis

One sample in the first ALT (n = 20) failed in 8,687 cycles. The confirmed value, based on the marketplace data, was 1.9. The shapes and locations of the failure in samples from the first ALT and the marketplace were similar (Fig. 5). The fracture of the suction reed valve came from its weak structure: (1) had an overlap with the valve plate, (2) used a weak material, and (3) had a sharp edge on the valve plate (Fig. 6).

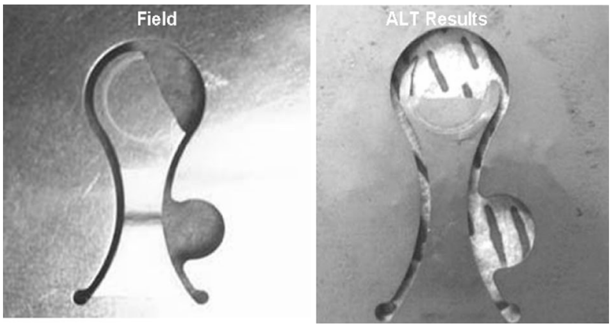


Fig. 5. Failure of suction reed valve in marketplace and 1st ALT result.

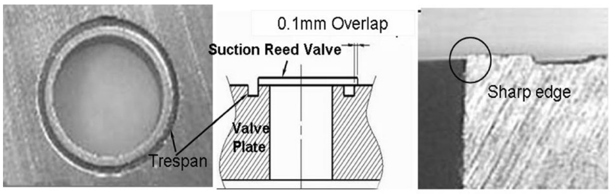


Fig. 6. Structure of suction reed and valve plate.

When the suction reed valve impacted the valve plate continually, it fractured easily. The dominant failure mode of the compressor was leakage and locking due to the cracking and fracturing of the suction reed valve.

It would appear that the ALT methodology was valid for reproducing the failure found in the field. First, the location and shape of the fractured suction reed valves from the field and those in the ALT results were similar. Fig. 7 represents the graphical analysis of the ALT results and market data on a Weibull plot.

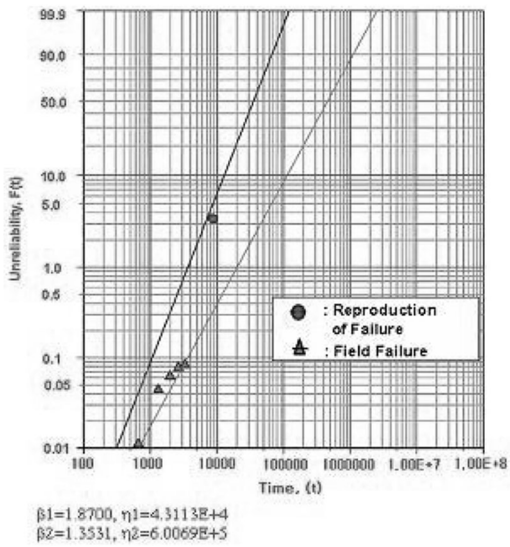
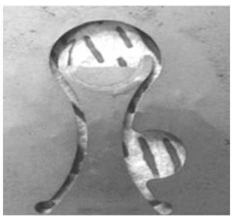
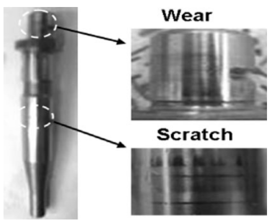


Fig. 7. Field data and results of accelerated life test on Weibull chart.

Table 3. Results of the ALTs.

	ALT(1)	ALT(2)	ALT(3)
	Initiai Design	Design Improvement (1)	Design Improvement (2)
In 23,000 Cycles, Crack & Fracture of Blade is less than 1	8,687 Cycles: 1/120 (5.0%) 8,687 Cycles: 19/20 Suspension	17,000 Cycles: 3/30 (10.0%) 17,000 Cycles: 27/30 Suspension	23,000 Cycles: 60/60 OK 29,000 Cycles: 60/60 OK
Suction Reed Vale Structure			
Material & Spec.	SANDVIK 20C 0.178t (Carbon Steel) → SANDVIK 20C 0.203t (Carbon Steel)	FCD500＋No Heat Treatment → FCD500＋Heat Treatment	SANDVIK 20C 0.203t (Carbon Steel) FCD500＋Heat Treatment

4.2 Parametric ALTs with corrective action plans and life prediction

Based on the first ALT and field data (Fig. 5), the AF and β values were 20.9 and 1.9. The test cycles and test sample number were calculated in Equation (14). One sample in the first ALT (n = 20) failed within 8,687 cycles. For the second ALT (n = 30), three samples were failed within 17,000 cycles. In the third ALT results (n = 60), the samples did not crack and fracture until 29,000 cycles of testing, as shown in Table 3.

The missing vital parameters in the design phase were trespan design of valve plate, material of the suction reed valve (Field and 1st ALT), and hardness of the crank shaft (2nd ALT). By the repetitive pressure loads, these design flaws may cause compressor to be locked.

The parameter design criterion of the newly designed samples was more than the target life, B_1, often years. The B_x life of the sample was calculated as:

$$B_x \cong \frac{h \cdot AF}{L_B} \cdot \left(\frac{x \cdot n}{r+1}\right)^{\frac{1}{\beta}} \tag{15}$$

The levels of the modified design parameters with corrective action plans included: (1) Trespan size, C1 from 0.73 mm to 1.25 mm; (2) Adding ball peening and brush process, C2; (3) Thickness of the suction reed valve, C3 from 0.178 t to 0.203 t (4) Adding tumbling process, C4 (Table 4 and Fig. 8).

The B_1 life of the compressor in the first and second ALTs was 1.5 and 2.5 years, respectively. Thus, the B_1 life of the newly designed compressor was 1.3 times that of the current one.

Table 5 shows the results obtained from the third ALT. The B_1 life of the redesigned compressor using Eq. (15) and Table 3 was 12.9 years. When the design of the current compressor was compared with that of the new one, the B_1 life expanded from 1.5 years to 12.9 years. Redesigning the valve plate and the suction reed valve and reinforcing the crank shaft were very effective in expanding the reliability of the newly designed compressor.

Table 4. Vital parameters based on the marketplace data and ALTs.

CTQ	Parameters			Unit
Locking	KNP	N1	Pressure diffierence	MPa
	KCP	C1	Trespan size	mm
		C2	Ball peening and brush process	-
		C3	Thickness of the suction reed valve	mm
		C4	Tumbling process	

Fig. 9 show the improved design of the compressor based on the ALT results.

5. Conclusions

To improve the reliability of the newly designed compressor in a refrigerator, we have examined the failure mode of the compressor and mechanisms and predicted its life using the accelerated life testing method. The following general conclusions were obtained:

1) Based on the returned compressors and 1st accelerated

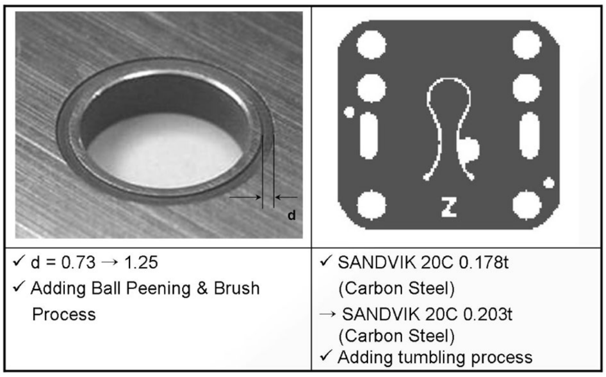


Fig. 8. Redesigned valve plate and suction reed valve.

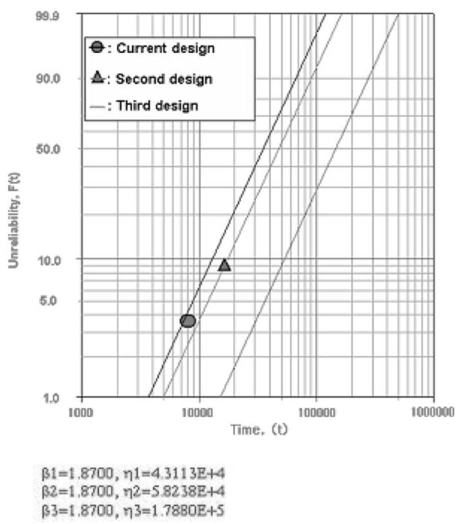


Fig. 9. Result of ALTs plotted in Weibull chart.

Table 5. Results obtained by the third ALT.

Factor	AF	β	h	r	LB	n
Values	20.9	1.9	29,000	0	357,700	60

life testing, the root causes of the failed suction reed valve in the compressor were (A) an overlap with the valve plate, (B) a weak material [SANDVIK 20C, 0.178t], and (C) the sharp edge of the valve plate. The design improvements, such as redesigned valve plate and suction reed valve, were effective for increasing the reliability of the reed valve.

2) In 2^{nd} accelerated life testing, the root causes of the failed compressor were (A) the wear of the crank shaft and (B) the interference between crank shaft and thrust washer. The design improvement was given to the heat treatment on the crank shaft.

3) After a sequence of reliability testing, the failure rate and the B_1 life of the redesigned compressor based were 0.06 percent per year and 12.9 years, respectively.

4) These methodologies - the inspection of the failed product, load analysis, and a sequence of ALTs - are very effective for improving its reliability.

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